TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS.

No. 312

HESSELMAN HEAVY-OIL HIGH-COMPRESSION ENGINE.

By K. J. E. Hesselman.

From "Zeitschrift des Vereines deutscher Ingenieure," July, 1923.

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In a Diesel engine, the fuel lying in front of the fuel valve is carried along by the cold blast air, which enters the cylinder at a velocity of about 300 meters (984 feet) per second, and is very finely divided. The high initial velocity is soon lost, owing to the resistance of the highly compressed air already in the cylinder, but nevertheless causes great turbulence. We may picture the further process as the entrance into the cylinder, at a high velocity, of clouds consisting of cold air and fine particles of fuel. On the surface of these clouds, the small particles of fuel come into contact with the hot air in the cylinder and are quickly heated, evaporated and ignited. The combustion, thus begun, is propagated into the interior of the clouds, assisted by the turbulence, but hindered by the low temperature of the blast air, which is still further cooled by its expansion from 60 to 35 atmospheres. Under certain conditions, this delay may be caused intentionally, in order to keep the expansion at constant pressure, or to avoid an explosive increase in pressure from the too sudden introduction of fuel. This generally results, however, in undesirable after-burning and increased fuel consumption.

* From "Zeitschrift des Vereines deutscher Ingenieure," July, 1923, pp. 658-662.

Too much fuel likewise delays combustion, since more heat is wasted in evaporating the larger amount of fuel, thereby lessening the increase in temperature and pressure. This explains why combustion often proceeds more smoothly when the engine is heavily loaded.

To what extent does the compressed blast air in Diesel engines accomplish its threefold task of introducing the fuel gradually into the cylinder, of finely spraying the fuel and of thoroughly and rapidly mixing it with the combustion air? Under carefully planned conditions, the first task has been gratifyingly, though by no means perfectly, accomplished. Under a small load, the correct adjustment of the fuel is difficult and under a suddenly changing load, difficulties also arise, due to the necessity of making corresponding changes in the pressure of the blast air. The task of spraying is admirably performed by compressed air, which also suffices for intimately mixing the fuel and air, although the low temperature of the blast air occasions certain disadvantages and the lively turbulence of the cylinder contents increases the amount of heat imparted to its walls.

Though fuel injection with highly compressed air is generally efficient and reliable, as demonstrated by years of use, it still has certain defects, in the elimination of which improvements will doubtless be made. The engine and its operation would be simplified, however, by dispensing with the compressor. Above all, it is important to simplify the mechanical operation and to

eliminate the cooling effect of the blast air.

Since solid injection of the fuel saves the work of the compressor, the mechanical efficiency of the engine is improved. It has not always been taken into consideration, however, that a portion of the compressor work is recovered through the expansion of the blast air in the cylinder, so that this improvement in the efficiency does not generally amount to more than 3 or 4%. The useful work, performed by the blast air in the engine cylinder, results, moreover, in an apparently more favorable fuel consumption. Since this is generally taken as the gage of the efficiency of the combustion, we cannot, as has hitherto often happened, disregard the fact that, with the same fuel consumption per horsepower, the heat from the fuel is utilized from 3 to 4% better in a solid-injection engine than in an ordinary Diesel engine.

Fuel Pump.- The difficulties of solid injection (namely, to introduce the fuel into the cylinder gradually, without violent increase in pressure, and to mix it simultaneously and thoroughly with the air) can be best illustrated by a concrete example. The example chosen is the successful engine (Figs. 1-2) designed by myself, a large part of the results obtained with it being capable, however, of universal application. Worthy of note, among other things, is the construction of the fuel pump, which, even in engines with several cylinders, has only one piston p. This piston must therefore make as many strokes as there are ignitions in the engine. The pump has only one intake valve,

s, as likewise only one delivery valve, t, which opens into the distribution chamber. Here the cams n operate a like number of distribution valves f, which deliver the fuel to the different cylinders. The pump delivery is regulated through the intake valve s in such manner that the latter closes after the pump piston has traversed a certain portion of its course, as, e.g., when the eccentric e, which drives the pump, is in the position 1, Fig. 2. After the pump has delivered to the engine cylinder the amount of fuel required by the loading of the engine at the time, the intake valve opens again at the point 1". As soon as the intake valve closes, the fuel is subjected to the pressure existing in the passage r, which is reached, e.g., at the point 1'. Then, when the delivery valve t opens, the cam-controlled distribution valve f is already open and the passage is free through the delivery pipe o to the fuel valve b. When the delivery valve t opens at the point 1', the pressure is transmitted into the distribution chamber and thence through the distribution valve f and the pipe o to the valve b. The latter valve is opened by the pressure and admits the fuel into the engine cylinder. This process is represented diagrammatically by Fig. 3. The central vertical line represents the ignition dead center. The horizontal lines correspond to the angular motions of the crank-pin from right to left. On the lowest horizontal line, A represents the instant when the intake valve s closes and B the instant when it opens again. The middle horizontal line shows the opening and closing of the

fuel valve b at the points C and D. The top horizontal line shows the beginning of the combustion at E and its close at F. The crank angles δ and γ therefore represent the time intervals between the closing of the intake valve s and the beginning of fuel injection and between the latter and the beginning of the combustion.

It is very important to know these angles and the conditions which determine their magnitude. I determined the angle δ with the aid of an indicator (Fig. 4) invented by myself. This consists essentially of a small disk, placed directly under the fuel valve b, and a lever which transmits the motions of this disk to a stylus and records them on a drum driven by the engine shaft.

With this indicator I made many diagrams which served as the basis for subsequent calculations. The most important results are as follows. The size of the angle δ is not affected by the length of the delivery pipe. Its length is of but little consequence, since the pressure is transmitted through it with extraordinary rapidity. Furthermore, the portion of the angle δ, which corresponds to the compression in the pump, is extremely small, so that the angle is but slightly affected by the pressure at which the fuel valve b opens. The average velocity of the pressure wave from the instant the intake valve s closes to the instant the injection begins stands in a definite ratio to the speed of the pump piston at the instant the delivery valve

t opens. This ratio is very exact and the slight variations brought out by the calculations are ascribable to unavoidable observation errors. The speed of the pump piston at the given instant is proportional to the revolution speed. The time which corresponds to the angle δ is inversely proportional to the revolution speed. Since the distance traversed by the pressure wave is constant and is the product of the velocity and the time, it follows that the angle δ is constant and independent of the R.P.M. of the engine.

This result is of practical importance. It means that the fuel, even at different revolution speeds, should always enter the cylinder at the same position of the crank. This was confirmed by other experiments. The ignition point does not therefore require changing for different revolution speeds. The fact that the angle δ may be quite large is principally due to the fact that the pressure wave is delayed by sharp turns in the valves. The angle accordingly depends on the type of pump and fuel valve and can hardly be determined in advance, but rather by experimentation, which is comparatively easy and has to be done but once for each new type.

The crank angle γ, which corresponds to the time interval between the beginning of the fuel injection and the ignition, can be determined with the aid of the shifted indicator diagram (Fig. 5), if the angle δ and the crank position are known, at which the intake valve s of the fuel pump closes. It is very

important to keep this angle as small as possible, so that no considerable quantity of fuel can collect in the cylinder.

It is not so easy to find what conditions determine this angle, as for the angle δ. According to my experiments, these conditions include the degree of compression, to a certain extent also the temperature of the fuel, the method of introducing the fuel into the cylinder, the manner of spraying, etc. I found a minimum value of 2.5° for γ, corresponding to 1/700 s (290 R.P.M.) though ordinarily the angle is larger, corresponding to 1/500 - 1/400 s.

In Fig. 6 the ignition point is represented as a function of the number of holes in the burner, the ordinates being the angles $\gamma + \delta$. Since δ is invariable, only the differences occasioned by the retardation of the ignition appear. With a smaller number of holes the ignition occurs considerably earlier. Compression of the fuel at the beginning of the injection hastens the ignition.

The velocity with which the fuel is injected depends greatly on the pump drive. At first the pump was driven by an eccentric and the intake valve was closed after a certain portion of the stroke had been made. In Fig. 7, the plain line shows the lifts of the needle-valve, which were measured by means of an optical indicator, while the dash line shows the amounts of fuel corresponding to every position of the valve. These amounts do not stand in a constant ratio to the valve lifts, because the pressure exerted on the valve from the cylinder side decreases for a

small valve-lift and the injection of small amounts of fuel. The amount of fuel forced through the fuel valve b increases rapidly to its maximum value, accompanied by a corresponding increase of pressure in the cylinder. The eccentric drive also has the disadvantage that, as the result of imperfectly understood conditions, probably connected with resonance, pressure variations of 50 or more atmospheres may occur and greatly disturb the functioning. The eccentric drive was therefore replaced by a cam, so shaped that the speed of the pump piston is very low at first and then gradually increases. This method enabled an orderly functioning of the pump and rendered it possible to control the increase of pressure in the cylinder. Fig. 8 shows that the needle-valve opens slowly and closes suddenly.

Further experiments with the fuel pump dealt with the hydraulic resistance of pipes of different sizes to fuels of various viscosities and also the hydraulic resistance of the different parts of the fuel pump and fuel valve and the determination of the number of extremely small openings in the nozzles of the fuel valves.

Fuel Valve.- The fuel valve (Figs. 9-10), in its steel housing, consists of a high-pressure filter, a valve-spring, a needle-valve and a nozzle with a burner. The valve-spring is composed of double spring disks turned from a single piece and connected on their inner circumference by a cylindrical piece. Its elastic properties were computed by Prof. Karl Ljungberg,

("Teknisk Tidskrift, Mechanik," 1920, No. 4), who found, among other things, that the ratio between its external and internal diameter must be determined by a definite law, in order to obtain the maximum effect. The adjacent double disks are separated on their outer circumference by distance rings and are also connected by shrunk rings. The dead space between the parts of a double disk is filled with divided rings held together by means of an elastic steel wire ring. Any change in length of the whole spring is the sum of the corresponding changes in the component disks. The spring must be accurately made of very strong steel, within limits easily attainable, however, in modern machine shops. According to the tests, this spring offers the same resistance to bending as a cylindrical steel rod of 11 mm (0.43 in.) and has a tensile breaking strength of 250 kg (551 lb.). The tension of the spring is adjusted from without. The requisite spindle is surrounded by a lead packing which can be easily kept tight, since the spindle does not move. On the opposite end of the spring, the head of the valve needle is held by a shrunk ring into which the needle-valve is screwed. As soon as the fuel pressure exceeds the pressure of the valve spring, the valve opens and the fuel is forced through the nozzle and burner into the engine cylinder. From the valve-lift diagrams (Figs. 7-8), which were made with an optical indicator, it appears that the valve opened and closed with extraordinary accuracy and without oscillations, which, in conjunction with the constant pressure during

the whole process of injection (Fig. 5), speaks well for the manner of construction. Its advantages consist in the small weight of the valve; its freedom from mechanical friction (since the friction of the liquid, on account of its low velocity, is small); the absence of the drive, which allows the valve to be placed in the most suitable location with respect to the fuel injection; the simplicity of the fuel distribution; the possibility of installing the valve in any desired location. The whole valve is attached to the cylinder by means of an iron clamp and a screw, so that it can be very quickly removed by loosening the clamp and unscrewing the pressure pipe.

Spraying.- It was first intended to introduce the fuel in the form of a conical spray by means of the nozzle shown in Fig. 11. The first experiments in the open air demonstrated, however, the hopelessness of this method. Such a fine spray evidently had too little weight and momentum to penetrate highly compressed air. Moreover, the uniformity of the spray would be too easily disturbed.

The attempt was next made to employ, also in conjunction with an automatic valve, nozzles or burners with a number of small holes, such as are used in ordinary Diesel engines or in Vickers mechanically operated valves. It was shown that, with the right shape and size of the valve seat with respect to the tension of the valve spring, it was possible to find a method of construction, which would operate, within broader limits, inde-

pendently of the diameter of the hole in the nozzle and its modulus of outflow. This method requires the pressure in the cross-section of the valve seat to be greatly reduced. Exhaustive experiments have verified the correctness of the calculations made in this connection.

The difference between the pressure p_2 in the nozzle and the pressure p_3 in the cylinder, depends on the amount of fuel delivered by the pump per unit of time, as well as on the maximum available cross-section of the hole in the nozzle and of the corresponding modulus of outflow. The pressure p_1 in the fuel valve should remain as uniform as possible during the injection and only slightly exceed the pressure required to open the valve. This pressure is regulated by means of the valve spring. On the other hand, p_2 varies between the compression pressure at the beginning of the injection and a certain maximum and subsequently falls rapidly to the pressure in the cylinder, whereupon the valve automatically closes. For different types, the excess pressure $(p_2 - p_3)$ in the nozzle at maximum delivery, can vary between 80 and 200 atmospheres, without appreciable effect on the pressure p_1 in the valve or on the rate of inflow of the fuel.

In order to illustrate the spraying of the fuel from the nozzle, I allowed, in a series of experiments at different pressures, the fuel to pass through correspondingly small holes into the open air. At a low pressure, the fuel spreads out at first in a smooth flow, which does not separate till some distance from

the nozzle. With increasing pressure the smooth portion shortens until, at a certain pressure, the separation begins at the nozzle. The resistance encountered by such a stream increases with the density of the air. The resistance of the smooth continuous portion is less than that of the finely divided portion, so as to warrant the assumption that a completely divided spray would not possess sufficient penetrating power.

A computation of the motion of small spherical drops of fuel in air, under pressures of one and nine atmospheres at 15°C (59°F) (the density of the air in the cylinder during combustion) and under the approximately correct assumption that the resistance increases proportionally to the density of the air, gave the following results. The distance traversed by a drop of fuel in a given time interval t is $s = \ln (1 + v_0 k t) : k$ (in which v_0 is the initial velocity) and $k = \frac{\psi \mu g \times 1.5}{d \gamma}$, in which ψ is the coefficient of resistance, μ the density of the air, g the acceleration due to gravity, d the diameter of the drop and γ the specific gravity of the fuel. This computation gives the curves plotted in Fig. 12. The continuous lines correspond to 100 m (328 ft.) per second and the dash-and-dot lines to 200 m (656 ft.) per second.

Among other things, the curves show that a drop of fuel of 0.01 mm (0.0004 in.) diameter, entering air under normal pressure (1 atm.) with an initial velocity of 100 m (328 ft.) per second, has already lost its energy after 0.02 second at hardly 100 mm (3.94 in.) from the mouth of the nozzle, so that it is

nearly motionless. In the same interval of time the drop penetrates only 15 mm (0.59 in.) into air under 9 atm. of pressure, thereby losing nearly all of its initial energy. Drops of 0.1 mm (0.004 in.) and even 1 mm (0.04 in.) diameter lose very much of their velocity during this short time. Even twice as great initial velocities do not materially increase the distances, because the resistance increases as the square of the velocity.

In the nozzles of fire hose it has been found that the carrying distance of the stream, under the same pressure and hence the same initial velocity, depends largely on the diameter of the nozzle and that the carrying distance is only slightly increased by increasing the pressure above a certain limit. We may therefore assume that the depth to which the compressed air is penetrated by the injected fuel increases up to a certain pressure and with decreasing diameter of the nozzle, beyond which, however, the distance penetrated will again diminish. The maximum attainable depth may be designated as the maximum penetrating power of the stream. It varies also with the amount of fuel injected per unit of time.

For the task of injecting fuel jets into the compressed air of the combustion chamber,, which must penetrate to a depth depending on the shape and dimensions of this chamber, there is no advantage in dividing the spray into very many jets, since each jet must have a certain cross-section and convey a certain amount of fuel. Neither is it of any use to raise the pressure in the nozzle above a certain point or (what amounts to the same thing)

to reduce the size of the holes in the nozzle below a certain limit. I have secured the best results with a five-hole burner, independently of the cross-section. It can hardly be accidental that Vickers also seems to prefer this number of holes. In ordinary Diesel engines I have had a similar experience, obtaining the best results with a burner plate containing eight holes. All experiments with a larger number of holes resulted in a larger fuel consumption.

Although a certain size of hole corresponds to the minimum fuel consumption, considerable variations of size in either direction only slightly affect the fuel consumption. This is of practical importance, because small variations in the size of the holes cannot be avoided.

There is still another argument against making the spray too fine. If an endless screw is inserted in the nozzle of a fuel valve, an extremely fine spray is obtained, which naturally appears very favorable for perfect combustion. With such finely divided fuel I have, however, never obtained perfectly smoke-free combustion, but always bluish exhaust gases. I surmise that a large portion of the fine fuel spray cannot penetrate the highly compressed air and remains near the burner where there is not enough air.

<u>Actual Process</u>.- This is not so simple. The problem is not only to obtain a fuel jet of sufficient penetration, but also, and chiefly, to bring about the complete combustion of the requi-

site amount of fuel within the shortest possible time.

When the fuel passes through the small openings into the air in the cylinder, it has a low temperature and a high velocity. Moreover, it offers the hot air only a relatively small surface area for the reception of its heat. The greater its distance from the nozzle, the more finely it is divided and the greater surface area it has. The heat is chiefly absorbed where its velocity is the lowest. The evaporation and combustion take place therefore, for the most part, at a quite definite distance from the burner, to which the shape of the combustion chamber must correspond (Fig. 13). The combustion chamber is bounded at the top by the lower side of the cylinder cover and at the bottom by the partially conical piston-head, whose apex lies directly beneath the burner. The lowest portions of the fuel jets then sweep the piston-head in its highest position.

When sprayed into the open air, the fuel forms jets whose outermost rays inclose an angle of about 15° (Fig. 14). It may be safely assumed that this angle is not greatly increased by spraying into compressed air. Hence, if the burner has five holes, the combined fuel jets, as seen from above, cover less than one-fourth of the piston-head. Consequently, the fuel can not automatically come into contact, in the short time available, with the amount of air required for complete combustion. An ideal solution of this problem would seem to be to rotate the burner, during the combustion, in such a way as to bring the fuel constantly into contact with unused air. This method, however, en-

counters serious difficulties. I, therefore, tried the reverse process and caused the air to circulate about the axis of the cylinder. In a four-stroke engine this result can be most easily obtained by admitting the air through a shielded intake valve approximately tangential to the circumference of the cylinder (Fig. 15). A decided circular motion, in addition to the eddies, is thus produced. During the compression the eddies mostly disappear, leaving a smooth circular motion, whose velocity depends on the position of the inlet valve. Carefully executed experiments have shown that a certain circular velocity is necessary to obtain the minimum fuel consumption. This most favorable velocity also depends on the number of holes in the burner and is smaller for burners with more holes. Apparently the velocity must be great enough for the air, during the injection, to describe an angle equal to that between two adjacent jets. Since the injection period is equal to about 0.1 of a revolution, the air in the cylinder must revolve twice as fast as the engine. In doing this, the circumferential velocity of the air remains within moderate limits. In my experimental engine, e.g., it may be estimated at about 8 m (26 ft.) per second.

The importance of having the right circumferential velocity of the air is clearly shown by Fig. 16, which shows a very definite minimum. If the air is admitted into the exact middle of the cylinder, the fuel consumption is increased 15%. At the maximum circumferential velocity an increase of 30% in the fuel con-

sumption was obtained. In both the extreme cases the exhaust gases had a dark appearance. The circular velocity has no effect on the ignition point, but probably does have on the form of the diagram, since the combustion line falls rapidly at a high circular velocity, even before the injection is finished. Although the most favorable circular velocity can hardly be determined otherwise than by experiment, only one experiment is, however, required for any given engine.

The accurately measured circular velocity of the air and the peculiar shape of the combustion chamber are the most important characteristics which, in the type of engine proposed by me, cause the rapid mixing of the fuel and air and contribute to its ability to stand high loading. This method seems to accomplish its purpose far better than the compressed air in Diesel engines. In ordinary solid-injection engines, there are only two motions which facilitate the mixing of the fuel with the air, namely, the motion of the fuel from the middle of the cylinder obliquely toward the piston-head and the downward motion of the piston. In my engine there is a third motion, the circular motion of the air, and it is apparently of considerable importance that these motions are perpendicular to one another. The changes undergone by a drop of fuel in the combustion chamber may be conceived as follows. As it leaves the injection nozzle, the drop has high velocity and low temperature. If its size exceeds a certain limit, it will be broken up by the resistance of the com-

pressed air. While moving toward the circumference of the cylinder, its velocity decreases and its outer particles become heated and evaporate. The resulting gas envelope is torn away by friction with the air, while the nucleus of the drop moves on until completely burned, which happens before it can reach the outer edge of the piston-head. The circular motion of the air now generates a certain motion of the fuel with relation to the air, even in the outer portion of the combustion chamber where the drop of fuel has practically lost all its initial velocity. After the combustion has actually begun, it proceeds rapidly and is very complete, as is shown by the rapid falling of the expansion line immediately after the injection, by the low final pressure and by the low temperature of the exhaust gases. The final pressure in the experimental engine, under full load and at $p_m = 5.35$ atm., was only 2.1 atm., or about 0.5 atm. less than in an ordinary Diesel engine.

The energy consumed in producing the circular motion of the air does not exceed 1% of the engine power. The heat losses, which in an ordinary Diesel engine cause violent eddies, are probably smaller in my engine, because the circular velocity of the air is relatively small and because the same portions of the air generally come into contact with the cooled walls.

The engine functions, like most solid-injection engines, with relatively low preliminary compression and according to the uniform-pressure combustion with preliminary explosion, corre-

sponding to the English "dual combustion" in which a portion of the fuel burns at approximately constant volume, thus raising the pressure and temperature in the combustion chamber and favorably affecting the succeeding combustion at approximately constant pressure. The indicator diagram accordingly occupies an intermediate position between the one for the Diesel engine and that for the gas engine.

The compression is so adjusted that reliable ignition is obtained, even in starting the cold engine. It can be considerably lower than in an ordinary Diesel engine, in which the blast air has a cooling effect and renders ignition difficult. My engine runs more easily and smoothly with only 28 atm. compression than a similar Diesel engine with 35 atm. Larger engines would probably function satisfactorily with only 25 atm. and perhaps still smaller pressures in warm regions.

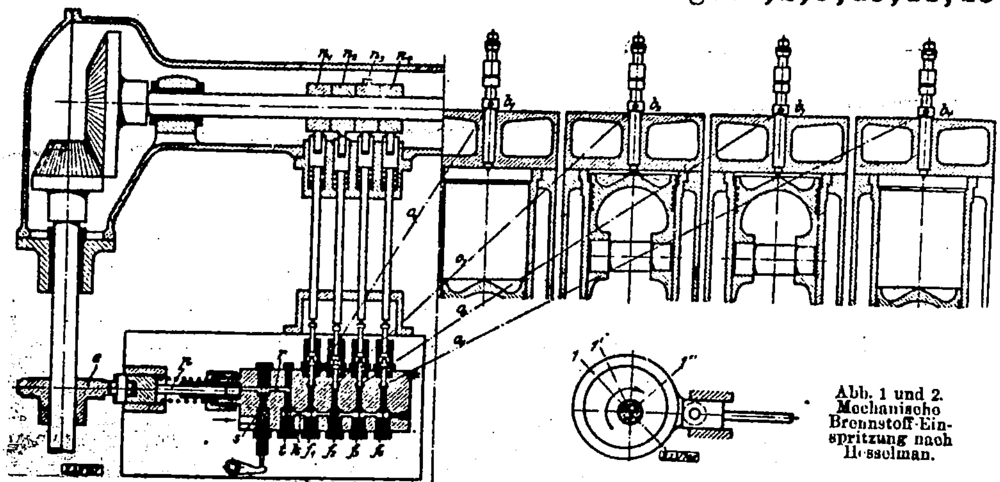
Experiments with higher compression have shown no appreciable saving in fuel. This is perhaps due to the fact that the theoretically greater advantages of higher compression are counterbalanced in the fuel injection by the disadvantages of higher density and greater resistance of the air in the cylinder.

The maximum pressure of combustion bears a certain relation to fuel consumption, in so far as the smallest fuel consumption corresponds to a definite maximum pressure. At a higher pressure, perhaps the improvement in thermal efficiency is offset by a loss in the mechanical efficiency. The fuel consumption is, however,

only slightly affected by a difference of a few atmospheres in the maximum pressure.

The ignition point has but a slight effect on the fuel consumption. Hence it is only necessary to adjust the ignition point so that the engine will run smoothly. In the new engine this adjustment is made as easily as in an automobile engine. As in the Diesel engine, the ignition point requires attention only in so far as it affects the maximum pressure.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.



Figs. 1 & 2 Hesselman system of mechanical fuel injection.

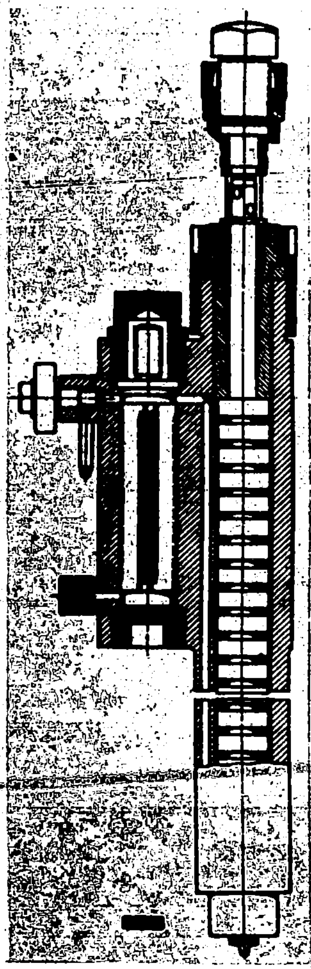


Fig. 9 Fuel valve

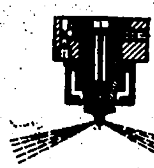


Fig. 10 Lower end of fuel nozzle

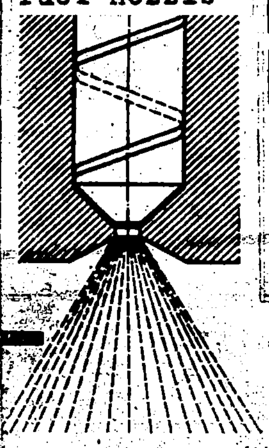


Fig. 11 Nozzle for conical spraying

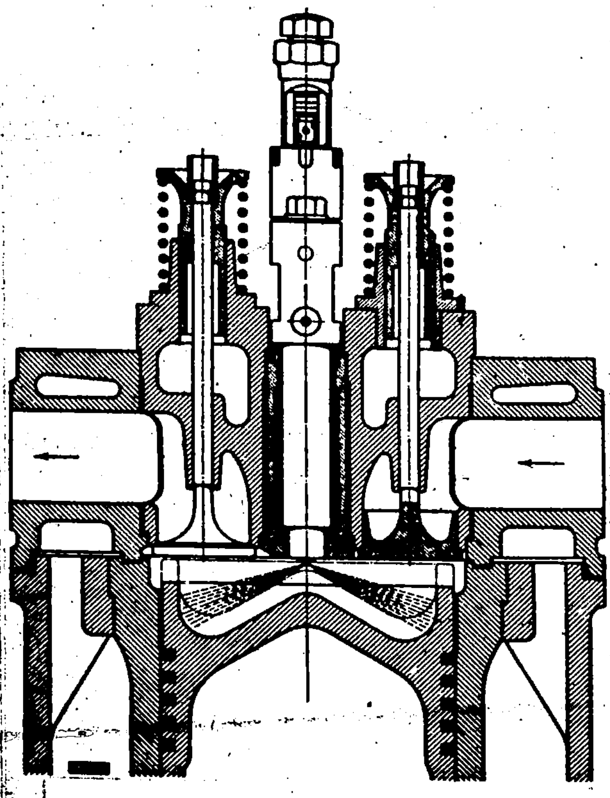


Fig. 13 Cylinder head

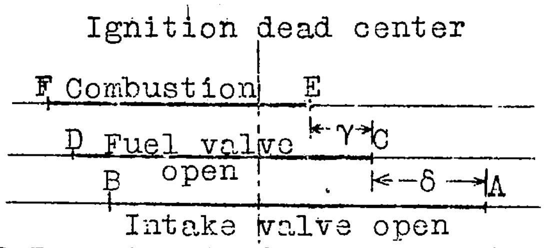


Fig.3 Ignition dead center. Retardation of injection and ignition.

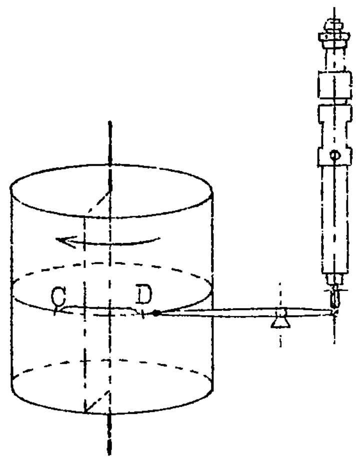


Fig.4 Indicator apparatus.

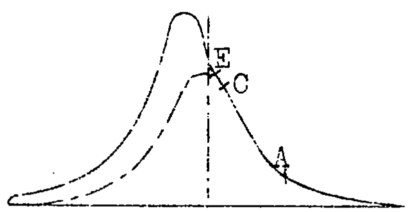


Fig.5 Shifted indicator diagram.

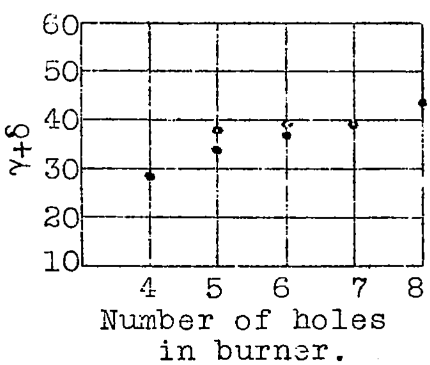


Fig.6 Retardation of ignition.

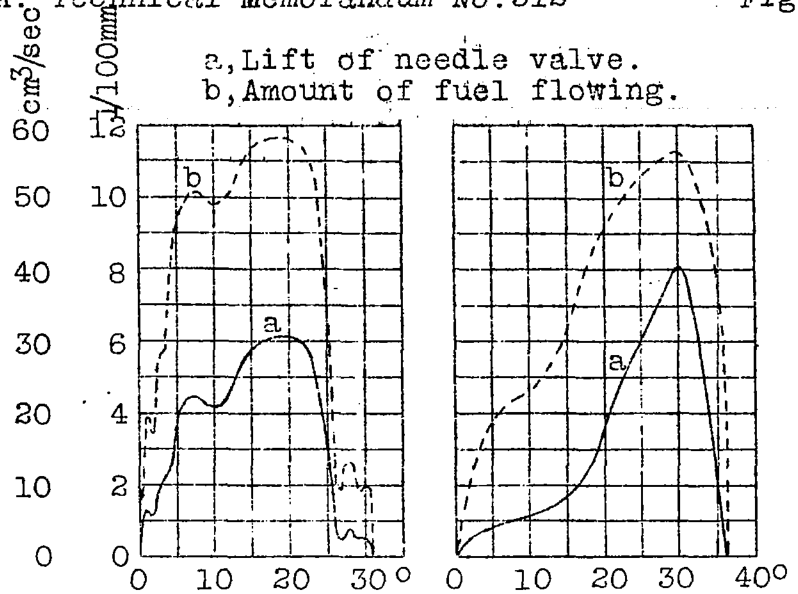


Fig.7 Crank drive. Fig.8 Cam drive.
Effect of pump drive.

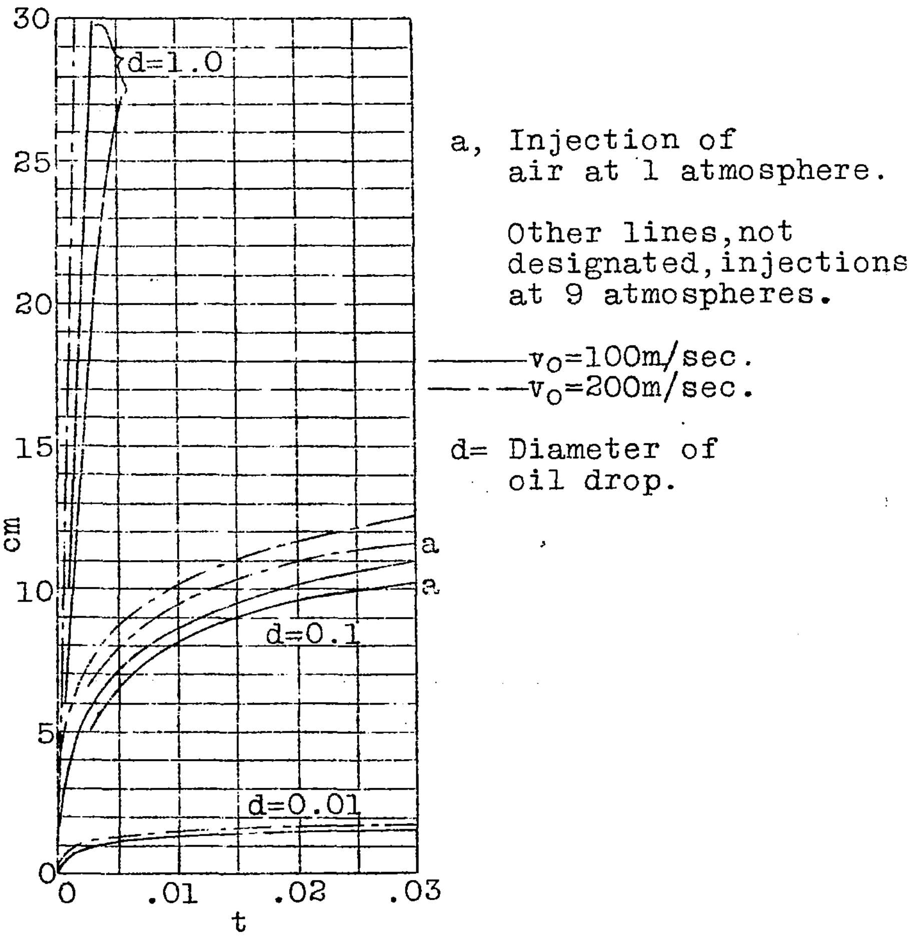


Fig.12 Penetration of oil particles into air at various densities and initial velocities.

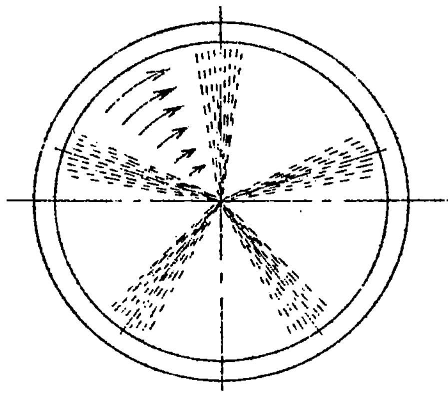


Fig.14 Fuel atomization in cylinder.

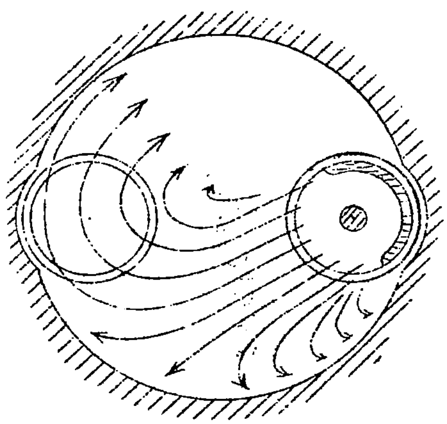


Fig.15 Introduction of air into cylinder.

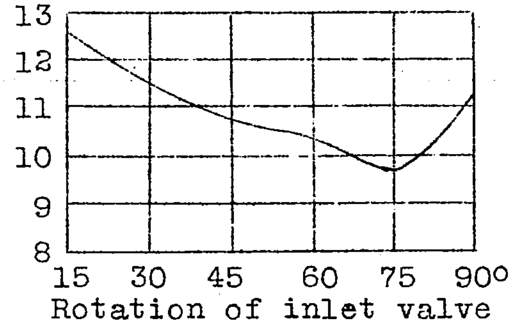


Fig.16 Relative fuel consumptiom at various adjustments of inlet valve.